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A study of exhaust gas temperature measurement methods

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Cambridge, Massachusetts; Massachusetts Institute of Technology

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A STUDY OF EXHAUST GAS TEMPERATURE MEASUREMENT METHODS

by

Lieut. Comdr. T.M. Bennett, USN
Lieut. Comdr. R.T. Boyd, Jr. USN
Lieut. Comdr. F.P. Cuccias, USN
Lieut. Comdr. G.C. Duncan, USN

M.S. Degree MIT 1947

Thesis
B38

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A STUDY OF EXHAUST GAS
TEMPERATURE MEASUREMENT METHODS

by

Lieutenant Commander Thomas M. Bennett, U.S.N.
Lieutenant Commander Randall T. Boyd, Jr., U.S.N.
Lieutenant Commander Francis P. Cuccias, U.S.N.
Lieutenant Commander George C. Duncan, U.S.N.

Submitted in Partial Fulfillment of the Requirements for the
Degree of
Master of Science in
Aeronautical Engineering
from the
Massachusetts Institute of Technology
1947

Thesis
B38

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Cambridge, Massachusetts,
May 23, 1947.

Professor Joseph S. Newell,
Secretary of the Faculty,
Massachusetts Institute of Technology,
Cambridge, Massachusetts.

Dear Sir:

A thesis entitled "A Study of Exhaust Gas Temperature Measurement Methods" is herewith submitted in partial fulfillment of the requirements for the degree of Master of Science in Aeronautical Engineering.

ACKNOWLEDGMENTS

Rogowski, A. R. Massachusetts Institute of Technology.
Leary, W. A. " " " "
Ku, P. M. " " " "
Livengood, J. C. " " " "
Kano, C. " " " "
General Electric Company, Schenectady, New York.

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SUMMARY

An investigation was performed on a CFR, single cylinder, four stroke engine to compare three different methods of measuring the temperature of the exhaust gases. The three methods used were:

1. Computation from indicator card.
2. Direct reading of the G.E. shielded thermocouple placed at the exit of the exhaust port.
3. Computation by use of a heat exchanger.

The tests were carried out at three different compression ratios and through a range of five fuel air ratios.

The indicator card method produced the highest temperatures. The G.E. Thermocouple temperature showed an energy content of the exhaust gases lower than the energy content as computed by the indicator method by approximately 6% of the heating value of the fuel. The heat transfer method gave temperatures lower than the other two and approximately ten percent of the fuel heating value below the indicator method. Good correlation was obtained among the three methods in the temperature trends with the maximum temperature at approximately chemically correct fuel air ratio.

The results of these tests are shown as plots of temperature vs percent of chemically correct fuel air ratio. Final conclusions made are that the G.E. Thermocouple is the most practical within the range of the instrument and that the heat exchanger method can be made more accurate than in this study and can be used over a much greater range than the thermo-

couple. The indicator card method is not considered an actual temperature measuring device.

INTRODUCTION

The measurement of energy in exhaust gases has never been successfully determined due to the inherent difficulty of measuring temperatures in a hot moving stream. Since an accurate measurement of these temperatures is necessary for engine heat balance and for the design of auxiliary turbine units, it is considered extremely important that a method be devised which will give satisfactory results.

The present investigation is a comparison between two of the most satisfactory methods to date, the General Electric shielded thermocouple (Ref. 1), and High Speed Engine Indicator card, with the heat exchanger method of this thesis.

This work was done in the Sloan Automotive Laboratory at the Massachusetts Institute of Technology in April and May, 1947.

DESCRIPTION OF APPARATUS

GENERAL

The set-up and arrangement of the apparatus used in this investigation is shown in Figs. 1, 2, and 3.

ENGINE

The engine was a CFR, single cylinder, four stroke engine. It was liquid cooled, had single spark ignition, a variable compression ratio, a bore of 3.25 inches, and a stroke of 4.5 inches. It had single inlet and exhaust valves with no overlap. The engine drove, besides the dynamometer, the oil pressure pump, the ignition breaker points, the tachometer, the dynamometer exciter, and the M.I.T. high speed engine indicator.

INLET SYSTEM

Air supplied to the engine was passed through a sharp-edged orifice (0.515 inch diameter - installed with flange taps according to ASME specifications -- Refs. 2 and 3), a 50 gallon surge tank, a throttle valve, a vaporizing tank, and an intake pipe leading to the engine. Temperature of the inlet mixture was controlled by supplying either low pressure steam or cooling water to the vaporizing tank jacket.

FUEL AND FUEL SYSTEM

The fuel used was one hundred octane gasoline supplied by the laboratory system. The fuel was supplied by an externally driven fuel pump and passed through a bubble separator,

a rotometer, and a needle valve directly over the vaporizing tank. This valve was used to pressurize the supply line to avoid vapor lock. Vaporization was obtained by heating the vaporizing tank with steam (Fig. 4).

CYLINDER COOLING SYSTEM

Cooling was provided by circulating water through the water jacket by an externally driven circulating pump. The temperature was maintained constant at boiling point and excessive boiling was prevented by cooling coils.

LUBRICATING SYSTEM

The oil pressure was obtained through an engine-driven pump. The oil circulating pump was externally driven. The oil temperature was satisfactorily controlled by a heat exchanger using low pressure steam or water.

IGNITION SYSTEM

Ignition was provided by a breaker operating at one-half crankshaft speed. The electrical circuit is shown in Fig. 5.

EXHAUST SYSTEM

The exhaust gases were discharged from the engine into a 1-1/4 inch iron pipe expanding into a 2 inch pipe in which the General Electric shielded thermocouple was housed (Fig. 6). From here the gases were passed through a flexible steel coupling into a heat exchanger for cooling purposes. The gases from the heat exchanger discharged into a surge tank and thence exhausted to the trench. The entire system from

cylinder to discharge was lagged to reduce heat loss to a minimum. A detailed sketch of the exhaust system is shown in Fig. 3.

MEASURING INSTRUMENTS

A large electric cradle-dynamometer was used to measure torque (Fig. 7). Speed was determined by a calibrated mechanical tachometer. Air flow was measured with a standard sharp-edged orifice constructed in accordance with the ASME Fluid Meter Committee specifications. A differential water manometer was used to measure the pressure difference across the orifice. The orifice diameter was 0.515 inches with standard flange pressure taps. The calibration and corrections for temperature and pressure for this set-up are shown in Fig. 8. Fuel flow was measured with a Fischer and Porter Stabl-Vis Rotometer; calibration curves for this rotometer are shown in Fig. 9.

Atmospheric pressure was measured by a mercury barometer; inlet pressure was measured in the vaporizing tank by a water manometer, as was the exhaust pressure in the exhaust surge tank. Spark advance was measured by a neon light (excited by the spark discharge) which revolved with the crankshaft.

Cylinder inlet temperature was measured by a mercury bulb thermometer in the inlet pipe. Circulating water temperatures through the exhaust gas heat exchanger were measured by mercury bulb thermometers at intake and outlet. Exhaust gas temperatures were measured in four places; one, shortly after the exhaust port of the engine by the General Electric

shielded thermocouple, and at three places in the exhaust surge tank by iron-constantan thermocouples. Oil temperature was measured by an iron-constantan thermocouple in the crankcase. Fuel temperature was measured directly after the rotometer by a mercury bulb thermometer. The M.I.T. High Speed Indicator was used to obtain records of cylinder pressure versus crank angles. (Fig. 10). The M.I.T. transfer table was used to obtain pressure-volume diagrams therefrom.

PROCEDURE

The primary consideration of this investigation was to correlate the results obtained from the three methods used to measure exhaust gas temperatures. This was done by taking simultaneous readings of all the data necessary, as shown in Table I. The runs made were divided into groups of three different compression ratios with five different fuel-air ratios for each compression ratio. Compression ratios used were 7, 8, and 9, and fuel-air ratios used were 80%, 90%, 100%, 110%, and 120% of chemically correct fuel-air ratios.

The following conditions were held at constant values as indicated:

1.	Oil pressure	40±2psia.
2.	Jacket water temperature	212°F
3.	Inlet valve clearance	.012"
4.	Exhaust valve clearance	.014"
5.	Engine speed	1600±10rpm
6.	Spark advance	20°

The following factors were varied as necessary for test:

1. Fuel-air ratio
2. Compression ratio

Many familiarization runs were made to determine the best operating conditions. During these familiarization runs many difficulties were discovered which had to be overcome in order to proceed. First, the heat exchanger, used to cool the exhaust gases, was much more efficient than originally

designed (according to Ref. 4), and the cooler had to be decreased in size from 12' to 8', and from counter flow to parallel flow, to maintain a temperature well above 212°F in the exhaust surge tank. Also, due to excessive heat caused by lagging the exhaust pipes, a sodium-filled exhaust valve was used to maintain cooler valve temperature to lessen the probability of backfire.

DISCUSSION

In this thesis three methods of exhaust gas temperature measurements were investigated. Two methods, namely, determination of temperature from the indicator diagram and from the heat exchanger required computation and use of the Hottel Burned Mixture Charts for (CH_2) (Fig. 11). The third method was the direct temperature reading of the General Electric Thermocouple. A description of the use of the Hottel charts and the high speed indicator diagram for the solution of this problem is given below.

To compute the exhaust gas temperature from a high speed indicator card the First Law of Thermodynamics was used. From a point "X" on the expansion stroke line immediately before the exhaust valve opens an equivalent cycle was used through the blowdown and exhaust processes.

It was necessary to find a relation between the cylinder volume and the Hottel Chart volume, and to determine the percentage of residual gas "f" in the actual cycle. These two must be done together by trial and error.

Relation between V_{chart} and $V_{cylinder}$.

In actual cycle:

Fresh air/cycle = B pounds

Air in residual gas/cycle = Z pounds

Total air/cycle = (B + Z) pounds

Total fuel/cycle = (B + Z)F pounds, where F = fuel-air ratio

Total charge/cycle = (B + Z)(1 + F) pounds

Total residual gas/cycle = Z(1 + F) pounds

But $f = \frac{Z(1 + F)}{(B + Z)(1 + F)} = \frac{Z}{B + Z}$

So that $Z = \frac{Bf}{1-f}$

Total charge/cycle $= (B + Z)(1 + F) = (B + \frac{Bf}{1-f})(1+F) = \frac{B}{1-f}(1 + F)$

Chart mass $= (1 + F)$ pounds

Therefore $\frac{\text{chart mass}}{\text{cycle mass}} = \frac{1-f}{B}(\frac{1+F}{1+F}) = \frac{1-f}{B}$

And $\frac{V_{chart}}{V_{cylinder}} = \frac{1-f}{B}$

Thus a value for f was assumed and the $V_{cylinder}$ for point "X" converted to V_{chart}. This value and the pressure at point "X" were then used to locate the corresponding point on the Hottel Chart for the same fuel-air ratio.

Then expand adiabatically to the exhaust pressure to get the volume of the burned charge at exhaust pressure.

Now $f = \frac{V_{clearance}}{V_{exhaust}}$

This value of f was then compared with the assumed value, and the method of determining f repeated until the two values were equal.

Now the point "X" could be accurately located on the Hottel chart. By expanding to total cylinder volume converted to chart volume, and to exhaust pressure, two points "4" and "5" respectively, could be located on the Hottel chart.

Assuming that the chemical energy of the exhaust gas did not change during the exhaust process, the sensible enthalpy of the exhaust gas may be calculated as follows:

$$(1-f)H_{se} = E_{s4} - fE_{s5} + \frac{p_e(V_1-V_2)}{J}$$

and the temperature of the exhaust gas determined from the Hottel chart.

Determination of Exhaust Gas Temperature by the Heat Extraction Method

The heat extracted from exhaust gases (Btu/sec) by the water was found from the rate of water flow (lbs/sec) and its rise in temperature. Since the basis of the Hottel charts is one pound of air plus the necessary fuel to give the proper fuel air ratio, the Btu's/sec found from the water flow was converted to Btu's/lb air by dividing this value by the mass rate flow of air in lbs/sec.

Using the mean temperature of the gases in the surge tank, the enthalpy of these gases was determined from the Hottel chart at an entropy of 0.30. This entropy was used because the actual point was off the chart. This introduced no error because in this region enthalpy is a function of temperature only. By adding the heat extracted (Btu/lb air) the enthalpy of the gases at exhaust was found. From this value the temperature was read directly.

Design Considerations

The design of the heat exchanger for cooling was critical. The area had to be such that a reasonable rise in water temperature was produced (to minimize errors in reading); that the rate of water flow was sufficient to prevent

steam pockets; and that the temperature of the exhaust gases in the surge tank was sufficiently high to avoid water condensation.

Three ninety degree turns were made in the exhaust system before the heat exchanger. This was deemed necessary in order to prevent the shielded thermocouple from "seeing" the hot exhaust port or the entrance to the cooler. Also, these turns induced turbulence and helped to prevent stratification of the exhaust gases. To minimize the possibility of a large temperature gradient across the thermocouple, a third turn was introduced immediately before the cooler. This turn directed the exhaust gases vertically upward into the cooler which allowed easy removal of air from the cooling water system.

The heat exchanger consisted of a two-inch pipe concentrically placed in a three-inch pipe. One bend was made in the cooler making the discharge horizontal for convenience of installation and instrumentation. This was done by spider spacing around the inner pipe, tightly packing sand between the pipes, and cold bending the combination. This method proved very satisfactory and is recommended for any similar construction.

The surge tank was used to bring the exhaust gases to a sufficiently stagnant condition to allow accurate pressure and temperature measurements.

The whole system was lagged with high temperature insulation and it very effectively minimized heat losses to the

atmosphere. Due to this lagging, it is recommended that the exhaust pipes and flexible coupling prior to the heat exchanger be made of high temperature steel, if the use of a measuring thermocouple is anticipated. The use of a flexible coupling between the exhaust pipe and heat exchanger was considered advisable in this investigation to prevent damage to the engine cylinder head due to vibration between the engine mount and the heavy cooling system, and to allow for changing of the compression ratio without moving the heat exchanger.

RESULTS

The temperatures were computed according to the methods described in the Discussion and were plotted in Figs. 12-17. As can be seen from the curves, the trend of temperature change by each method agreed exceedingly well. Since a spark advance of 20 degrees was used in all runs, the shape of the curves for each set of compression ratios differed from one another. This was apparently due to the fact that best power spark advance was not used for each compression ratio and fuel air ratio.

The results from the heat exchanger method used in this investigation were not quite in agreement with the direct reading of the G.E. thermocouple and were well out of agreement with the indicator method. This was expected due to the simplifying assumptions made, namely, that the exhaust system process was adiabatic and the "blow-down" was isentropic.

The temperatures computed by the indicator method were fundamentally theoretical and are not to be considered the actual temperatures existing in the exhaust pipe. However, the nearer to this temperature that any method can come, the better the method. From the curves, the temperature difference between the theoretical and the computed temperature is on the order of 400°F. This is not considered to be close enough to give an accurate heat balance of the engine, nor to predict accurately the

energy remaining in the exhaust gases. No accurate heat balance has ever been performed on an internal combustion engine, but according to theoretical considerations, approximately 15% to 20% of the heating value of the fuel is dissipated in the cylinder and jacket. Because the exhaust pipe and port in this investigation were at such a high temperature, due to the lagging, it was expected that the heat transfer back to the cylinder from the exhaust pipe would appreciably increase the amount indicated above. Therefore, by taking an estimated 8% to 11% of the heating value of the fuel, additionally dissipated to the cylinder and jacket, and applying this correction to the curves already obtained by the cooler method, they can be made to fall directly on the indicator curves.

The G.E. thermocouple gave results that were closer to the indicator temperatures, the average error from this temperature being about 6% to 7% of the heating value of the fuel. Because of its small size and simple installation, this thermocouple was considered both practical and accurate. Therefore, it is concluded that this method is, to date, the most satisfactory method of measuring high temperatures in a hot moving stream within the limits of the instrument, that is, up to 1800 degrees Fahrenheit.

Since the error between the G.E. thermocouple reading and the heat exchanger method is only about three percent of the heating value of the fuel, and since this error can

be further reduced or eliminated by connecting the heat exchanger, with an insulating coupling, directly to the exhaust port of the engine, the method developed in this thesis is recommended for use with engines whose exhaust temperatures are both in and above the temperature range of the G.E. thermocouple. This is based on the assumption that the G.E. thermocouple gave results closest to the actual conditions.

CONCLUSIONS

1. Satisfactory correlation was obtained among the three methods investigated.

2. The General Electric shielded thermocouple gave values closest to the expected results and is the most practical reliable method.

3. The heat exchanger method is sufficiently accurate for the measurement of temperatures above the range of the G.E. thermocouple.

4. The indicator method can be used for rough results by subtracting approximately 6% of the heating value of the fuel from the energy of the exhaust gases determined by this method.

RECOMMENDATIONS

1. This investigation should be further carried on, connecting the water cooler (by insulated coupling), directly to the exhaust port to further reduce heat losses.
2. To establish a definite trend of the temperature curves, best power spark advance should be used for all runs.
3. If a similar set-up is used, high temperature metals should be used for the portion of the exhaust pipe between engine and cooler.

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6. "Temperature - Its Measurement and Control in Science and Industry", Reinhold Publishing Corp., 1941.

SAMPLE CALCULATIONS

Heat Exchanger Method

Conditions:	Compression ratio	8
	Fuel air ratio	120% of chemically correct

Mean temperature in exhaust surge tank (°F)	303
Mass rate of water flow (lbs/min)	3.43
" " " air " (lbs/sec)	0.0149
Rise in cooling water temperature (°F)	104

$$\text{Heat extracted by cooling water} = \frac{3.43 \times 104}{60} = 5.95 \text{ Btu/sec}$$

$$= \frac{5.95}{.0149} = 399 \text{ Btu/lb air}$$

Sensible enthalpy of gases @ 303°F (Btu/lb air)	102
Rise in sensible enthalpy of gases (Btu/lb air)	399
Final sensible enthalpy of gases (Btu/lb air)	501
Temperature of exhaust gases at exhaust (°F)	1555

Indicator Card Method

Conditions:	Compression ratio	8
	Fuel air ratio	120% of chemically correct

Symbols: M_a - mass of air (lbs/cycle)

V_{cyl} - actual cylinder volume (cubic inches)

f - mass fraction of residual gases

P_x - pressure at point "x" on indicator card

V_x - cylinder volume at point "x" on indicator card.

E_s - sensible internal energy (Btu/lb products)

H_s - sensible enthalpy (Btu/lb products)

$$M_a = \frac{\text{lbs/sec}}{\text{rps}/2} = \frac{.0149}{1593} \times 120 = .001121$$

Clearance volume = 5.333 cubic inches

Assume $f = .044$

From Indicator diagram

$P_x = 65.95\ \#/\text{in}^2 \qquad V_x = 37.73\ \text{in}^3$

$$V_{chart} = \frac{37.73}{1728} \times \frac{.956}{.001121} = 18.6\ \text{ft}^3$$

$$V_{clearance} = \frac{5.333}{1728} \times \frac{.956}{.001121} = 2.63\ \text{ft}^3$$

From Chart

$$V_5 = 60\ \text{ft}^3 \qquad \therefore f = \frac{V_{clearance}}{V_5} = \frac{2.63}{60} = .044$$

From Indicator diagram

$$V_4 = 42.664\ \text{in}^3$$

$$V_4 = \frac{42.664}{1728} \times \frac{.956}{.001121} = 21.05$$

From Chart

$$E_{s4} = 885 - 336 = 549$$

$$E_{s5} = 692 - 336 = 356$$

$$(1 - f)H_{se} = E_{s4} - fE_{s5} + \frac{P_e(V_1 - V_2)}{J}$$

$$= 549 - .044 \times 356 + \frac{14.7 \times 144 \times 18.32}{778}$$

$$= 583.6$$

$H_{se} = 610$ Btu/# air + fuel

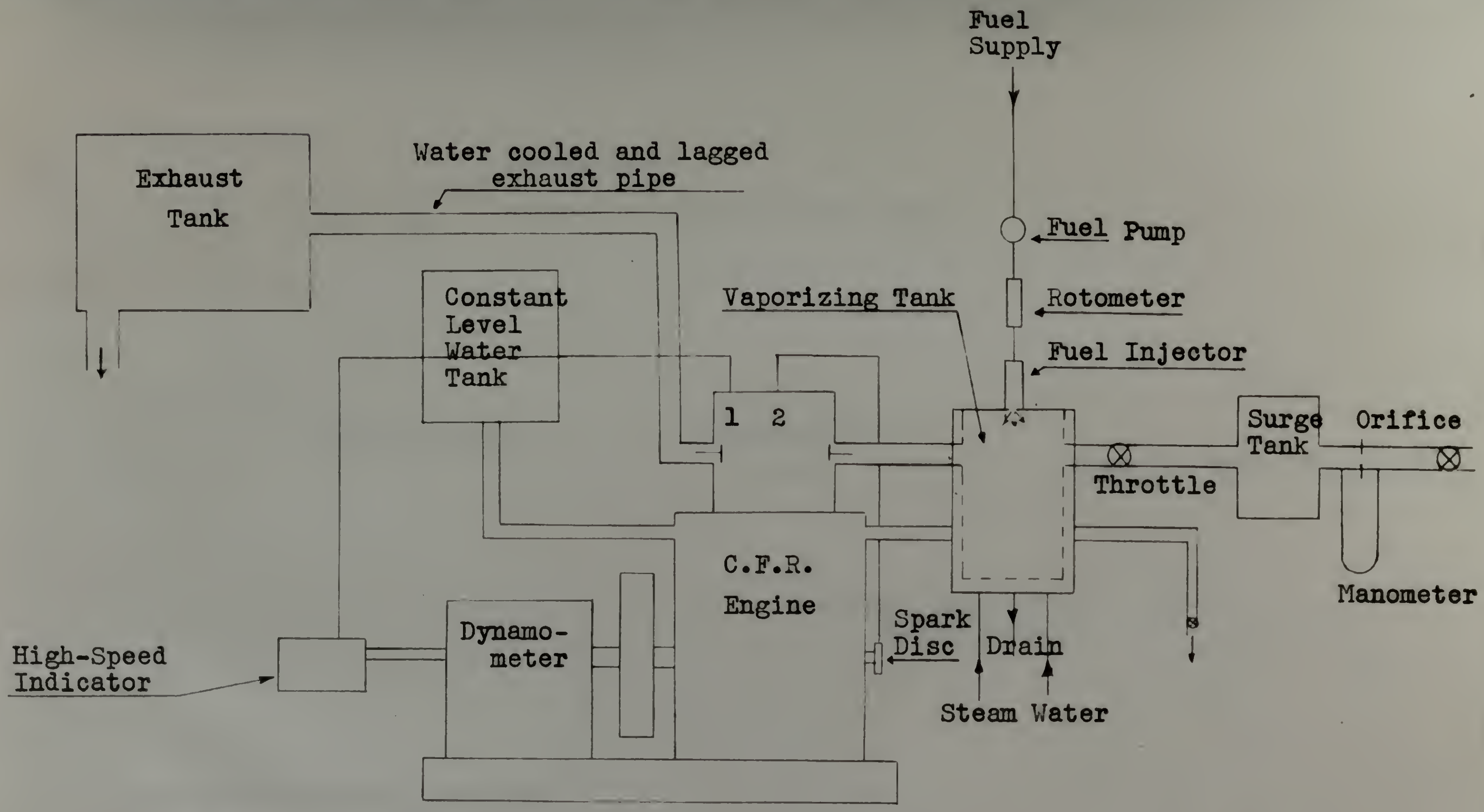
$T_e = 2345^\circ R = 1885^\circ F$

SLOAN LABORATORY

ENGINE CFR BORE 3.25" STROKE 4.5"

REMARKS	DATE	TIME	RUN	RPM	B.L.	COMP. RATIO	OIL PRES.	TEMP		P_1	P_E	T_1	AIR CONS.	FUEL CONS.	$\frac{F}{A}$	S.A.	AIR		FUEL ROTO.	WATER			THERMOCOUPLES			
								OIL	JACK.								P_1	ΔP		T_1	T_o	W	1	2	3	G.E.
					"Hg		PSI	°F	°F	"Hg	"Hg	°F	#/SEC.	#/SEC		°	"Hg	"H_2O	85°F	°F	°F	#/MIN	°F	°F	°F	°F
	4/12	1600	1	1610	7.25	7	40	160	212	28.30	29.92	141	.0150	.00120	.0798	20	29.92	12.30	9.90	54	170	2.95	330	325	337	1643
			2	1610	7.28	7	40	160	212	28.30	29.92	140	.0148	.00108	.0731	20	29.92	12.00	9.34	54	163	3.33	304	302	314	1744
			3	1610	7.10	7	40	160	212	28.30	29.92	155	.0148	.000983	.0665	20	29.92	12.00	8.85	56	174	3.38	328	323	338	1840
			4	1610	6.50	7	40	160	212	28.30	29.92	155	.0148	.000886	.0599	20	29.92	12.00	8.37	56	170	3.56	330	330	340	1820
			5	1610	5.60	7	40	160	212	28.30	29.92	155	.0148	.00079	.0532	20	29.92	12.00	7.85	57	173	3.51	338	332	348	1864
	4/24	1700	6	1593	6.95	8	38	136	212	28.11	29.73	150	.0149	.00119	.0798	20	29.73	12.14	9.85	56	160	3.43	300	300	310	1626
			7	1590	6.85	8	38	136	212	28.11	29.73	150	.0149	.00109	.0731	20	29.73	12.14	9.37	56	155	3.67	300	300	310	1705
			8	1587	6.60	8	38	134	212	28.11	29.73	150	.0149	.000991	.0665	20	29.73	12.14	8.90	56	157	3.71	301	301	311	1712
			9	1590	6.25	8	39	132	212	28.11	29.73	150	.0154	.000921	.0599	20	29.73	13.00	8.55	56	162	3.47	300	300	310	1706
			10	1590	5.40	8	39	129	212	28.11	29.73	150	.0156	.00083	.0532	20	29.73	13.30	8.07	56	175	3.09	303	302	313	1733
	4/24	2200	11	1595	8.60	9	41	126	212	28.11	29.73	134	.0156	.001244	.0798	20	29.73	13.30	10.08	55	162	2.90	279	278	288	1457
			12	1594	8.60	9	42	126	212	28.11	29.73	134	.0155	.001134	.0731	20	29.73	13.20	9.58	55	166	2.92	286	286	297	1546
			13	1594	8.50	9	42	125	212	28.11	29.73	129	.0156	.001039	.0665	20	29.73	13.30	9.14	55	176	2.80	300	299	310	1605
			14	1587	8.10	9	42	125	212	28.11	29.73	129	.0156	.000935	.0599	20	29.73	13.30	8.61	55	160	3.20	300	300	310	1565
			15	1577	8.10	9	42	125	212	28.11	29.73	129	.0156	.00083	.0532	20	29.73	13.30	8.06	54	158	3.25	299	299	309	1577

TABLE I



1. Indicator Cylinder Attachment
2. Spark Plug

SLOAN LABORATORY

EXPERIMENTAL C.F.R. ENGINE SETUP

Fig. 1

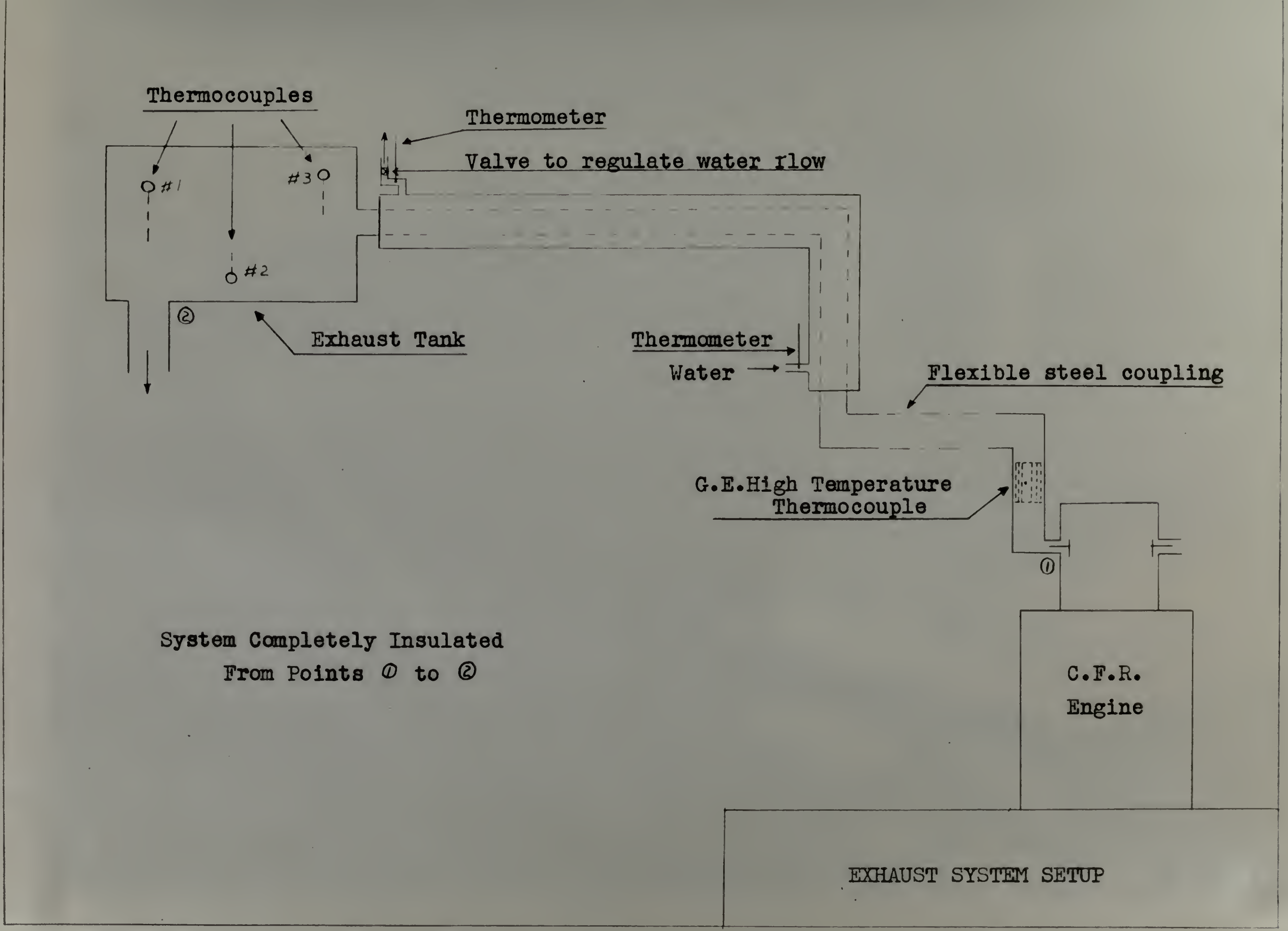

Thermocouples
Thermometer
Valve to regulate water flow
#1
#3
#2
②
Exhaust Tank
Thermometer
Water
Flexible steel coupling
G.E.High Temperature Thermocouple
①
C.F.R. Engine
System Completely Insulated From Points ① to ②
EXHAUST SYSTEM SETUP

Fig. 2

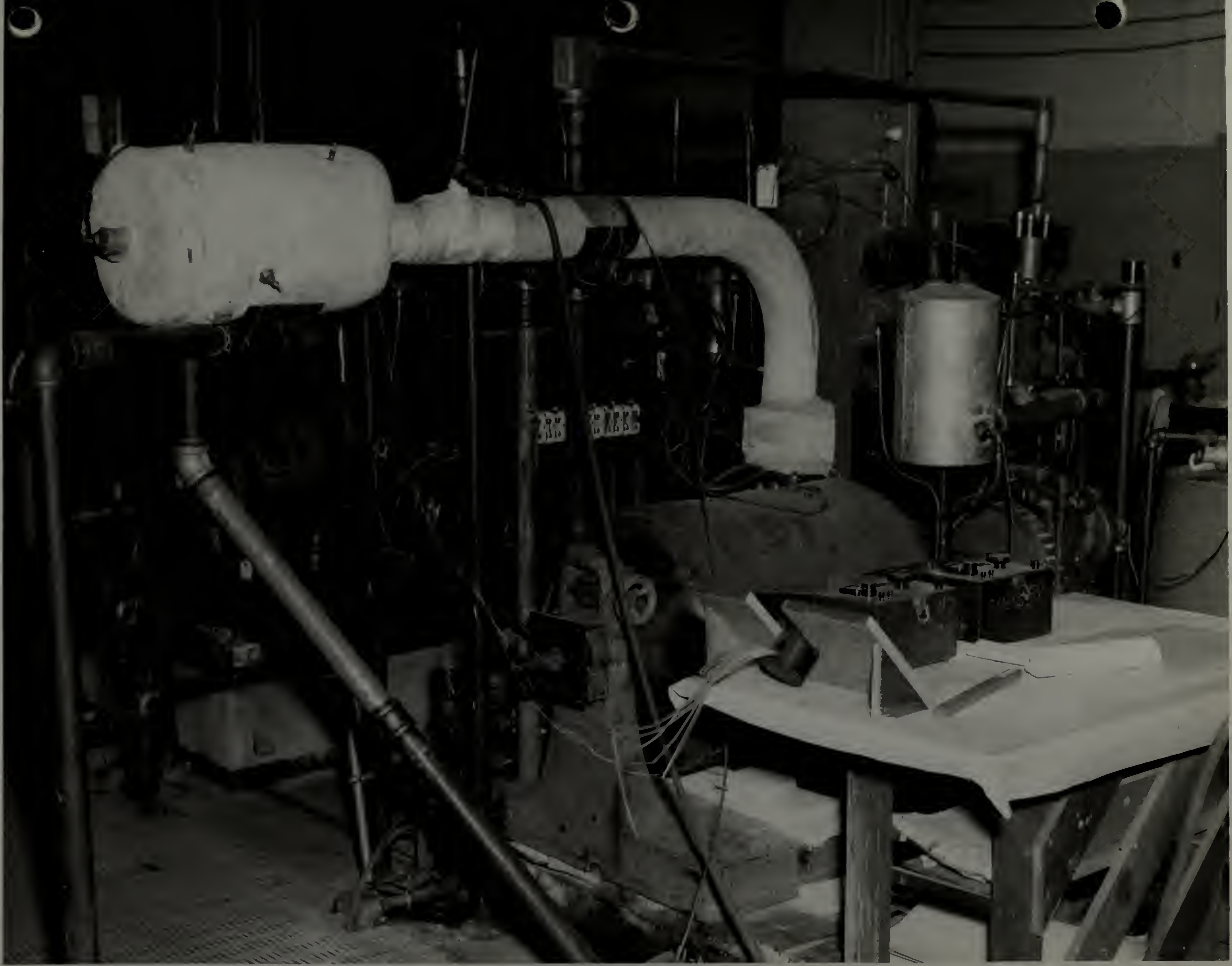
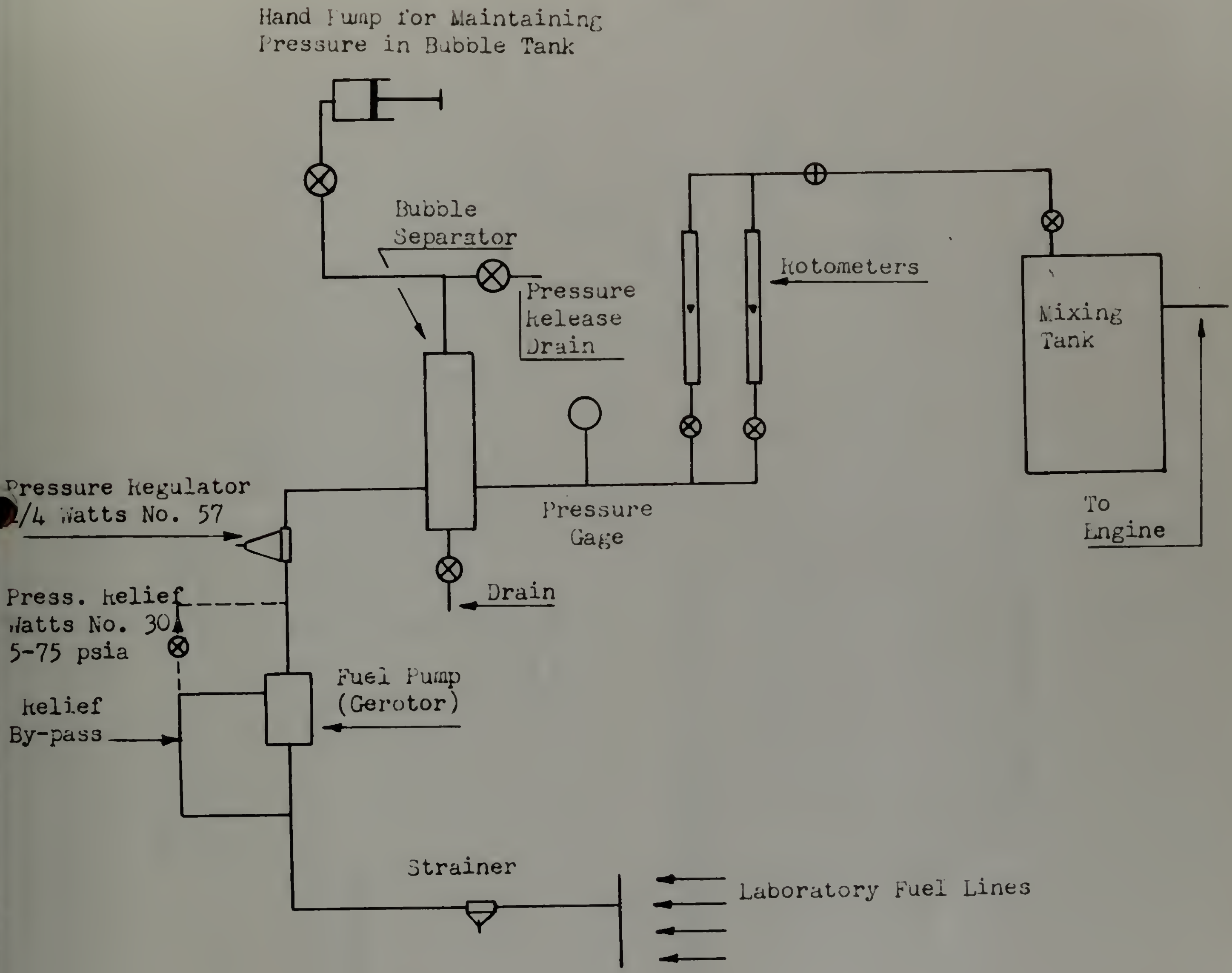
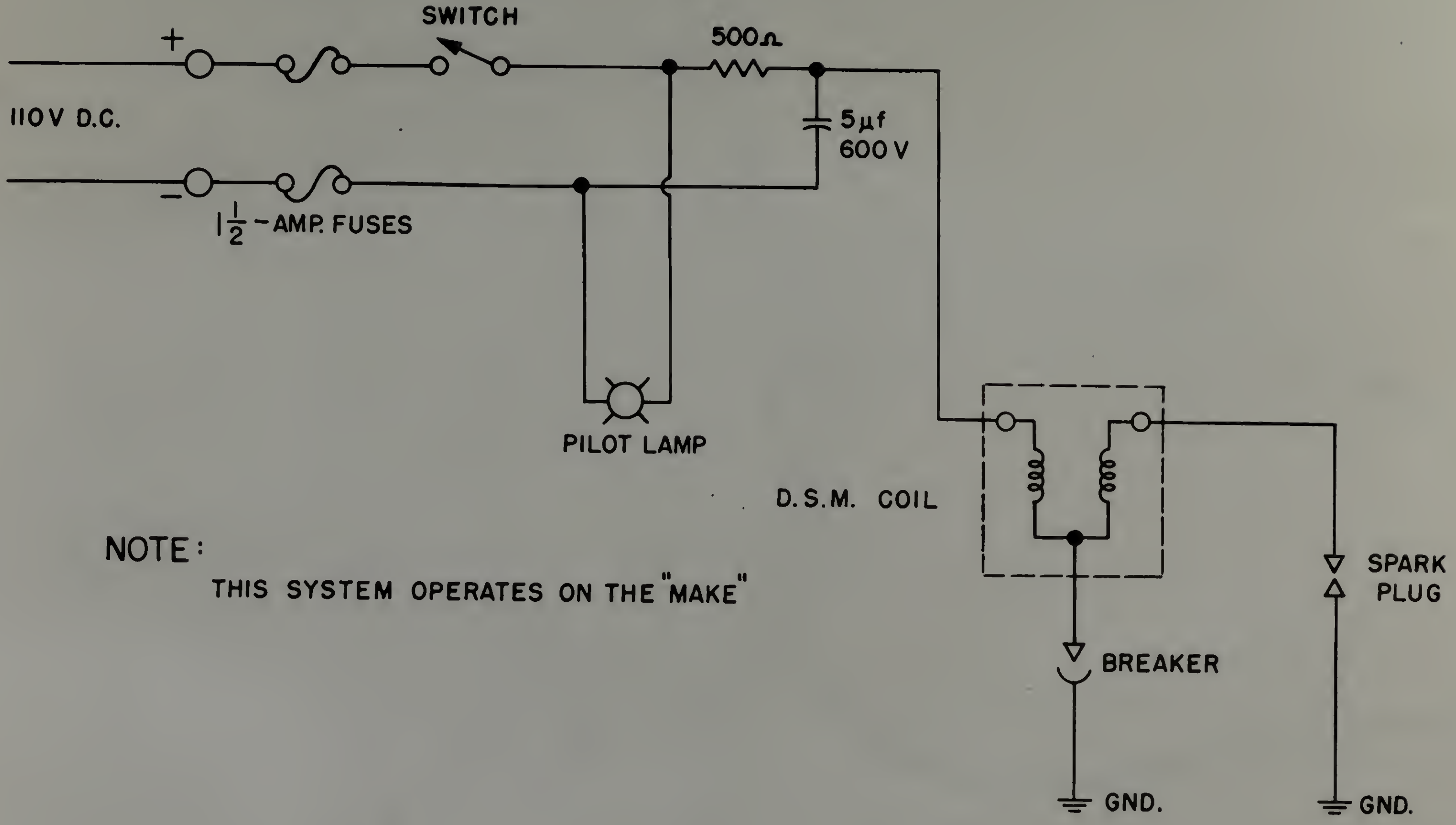


Fig 3



SCHEMATIC DIAGRAM OF FUEL SYSTEM

Fig.4



NOTE:

THIS SYSTEM OPERATES ON THE "MAKE"

WIRING DIAGRAM OF IGNITION SYSTEM

FIG. 5

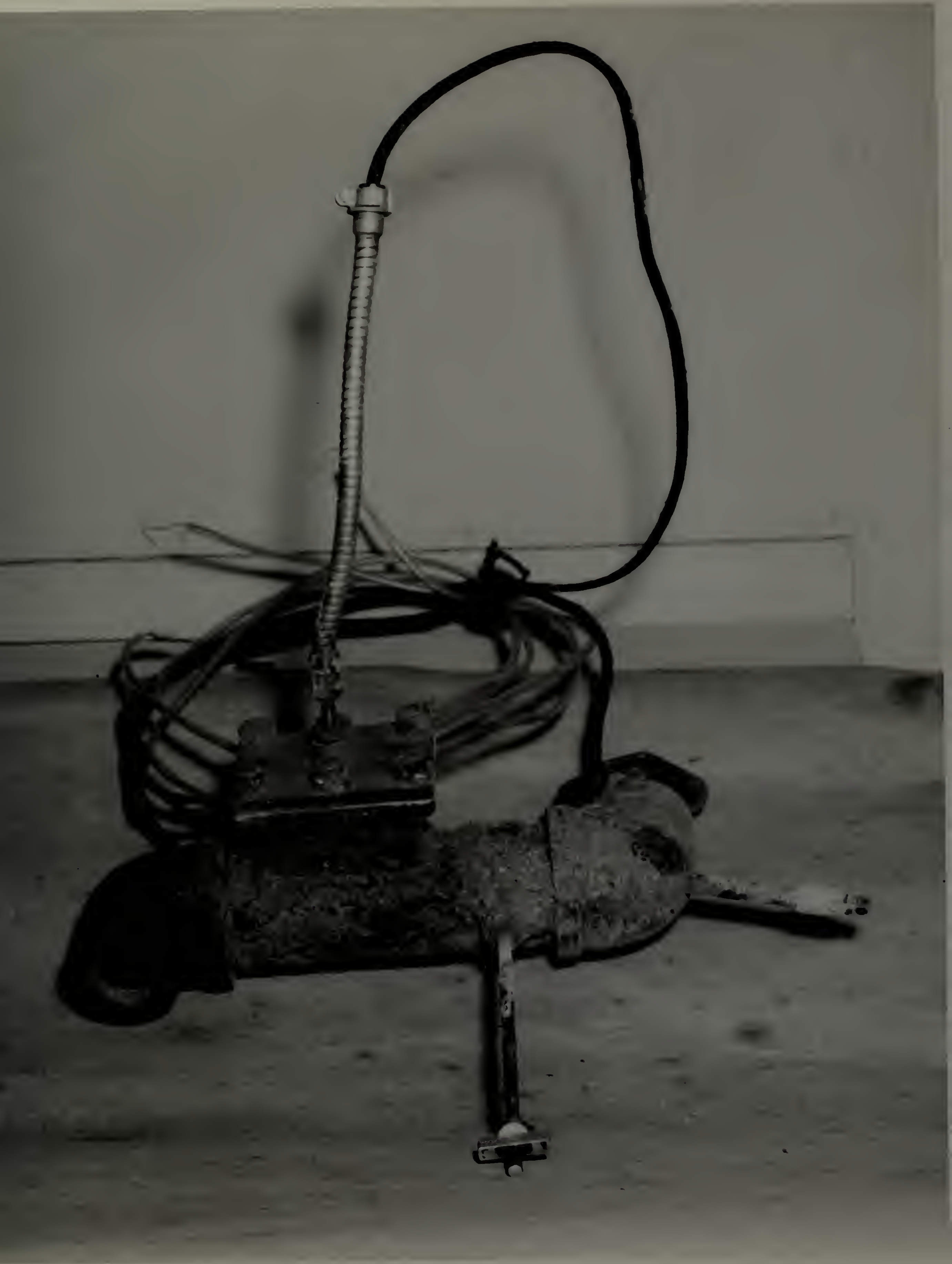
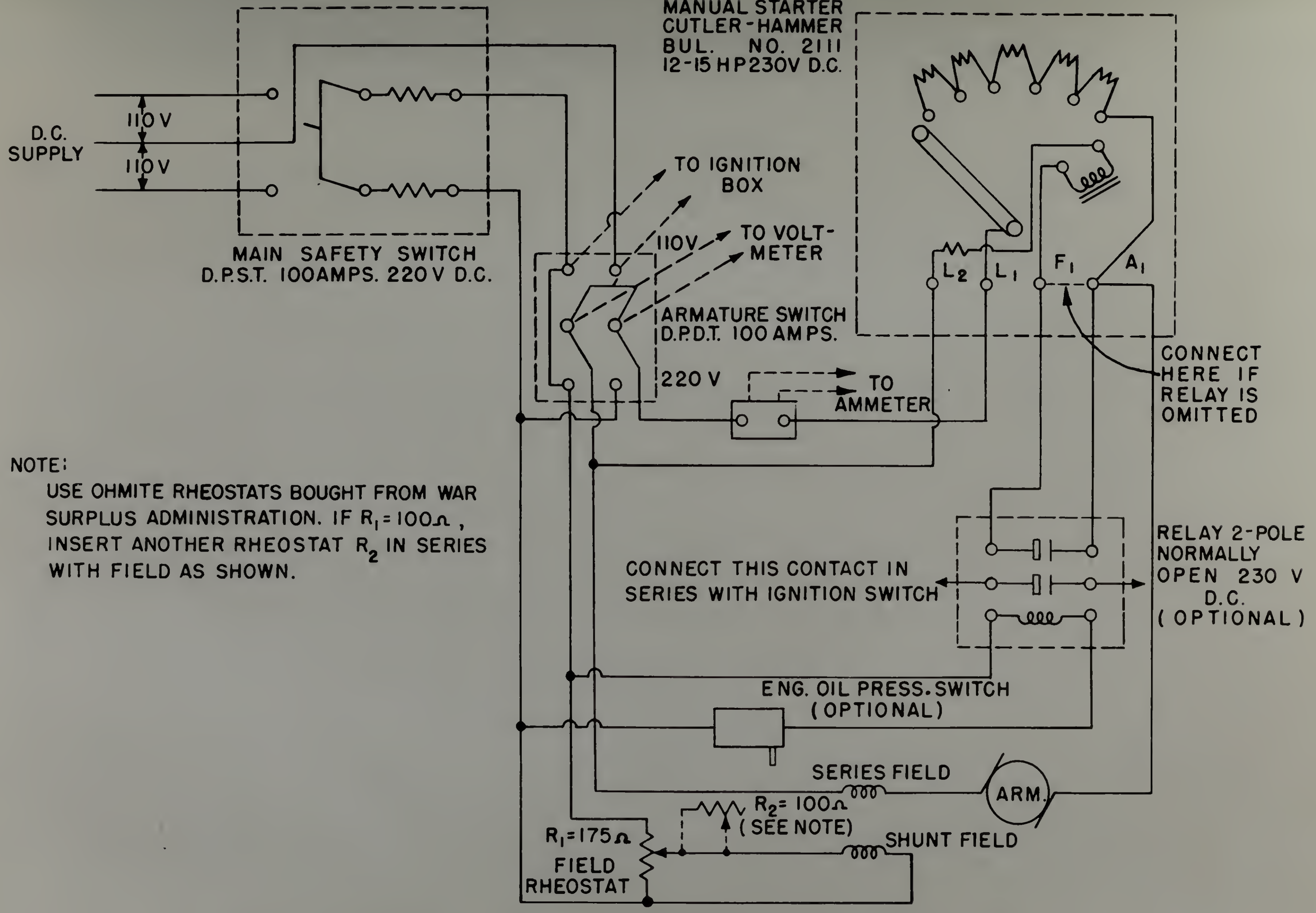


Fig. 6



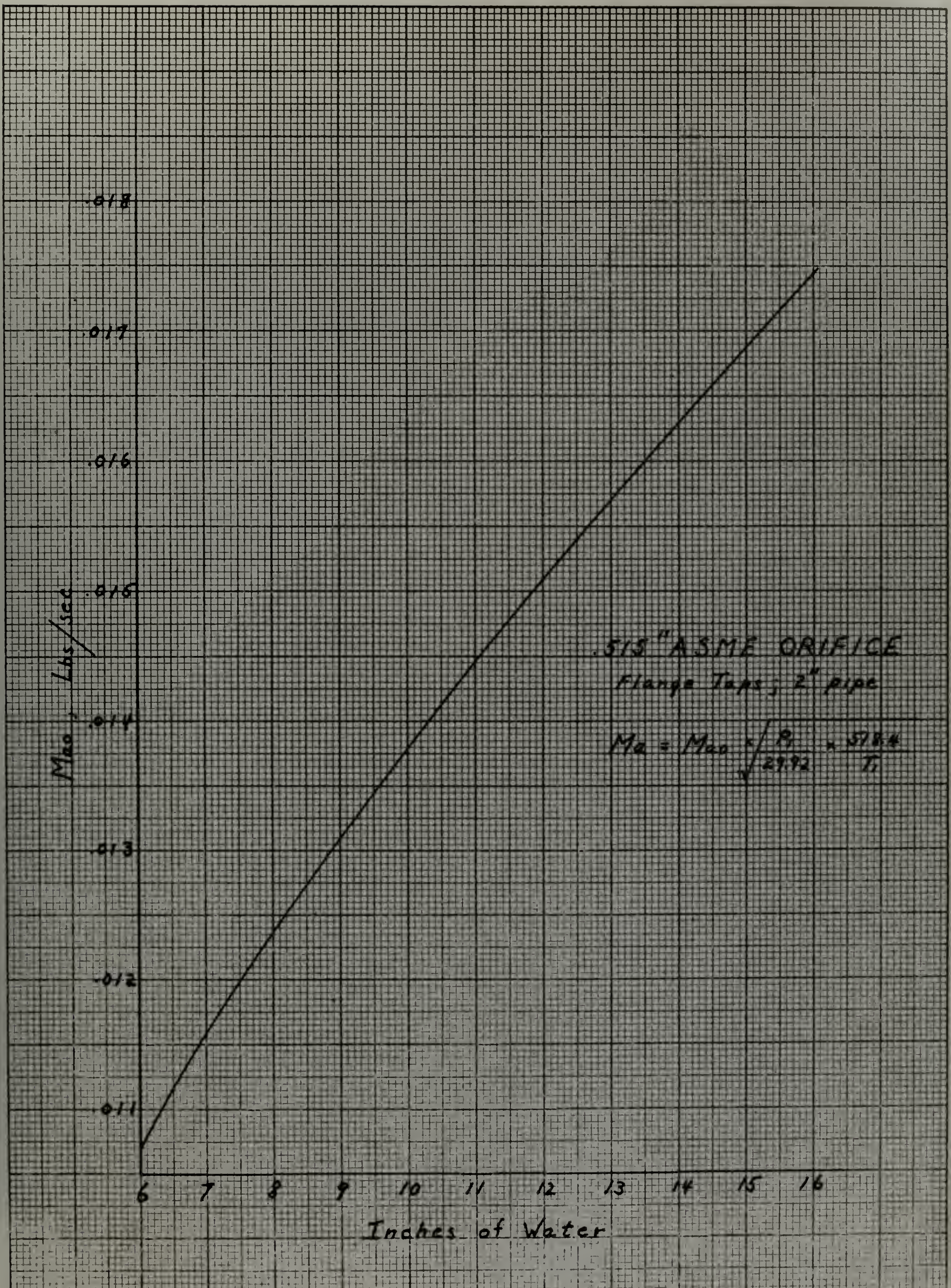
D.C. SUPPLY
110 V
110 V
MAIN SAFETY SWITCH D.P.S.T. 100AMPS. 220 V D.C.
MANUAL STARTER CUTLER-HAMMER BUL. NO. 2111 12-15 HP230V D.C.
TO IGNITION BOX
110V
TO VOLT-METER
ARMATURE SWITCH D.P.D.T. 100 AMPS.
220 V
TO AMMETER
L2
L1
F1
A1
CONNECT HERE IF RELAY IS OMITTED
CONNECT THIS CONTACT IN SERIES WITH IGNITION SWITCH
RELAY 2-POLE NORMALLY OPEN 230 V D.C. (OPTIONAL)
ENG. OIL PRESS. SWITCH (OPTIONAL)
SERIES FIELD
ARM.
R2= 100Ω (SEE NOTE)
SHUNT FIELD
R1=175Ω
FIELD RHEOSTAT

NOTE:

USE OHMITE RHEOSTATS BOUGHT FROM WAR SURPLUS ADMINISTRATION. IF $R_1 = 100\Omega$, INSERT ANOTHER RHEOSTAT R_2 IN SERIES WITH FIELD AS SHOWN.

WIRING DIAGRAM OF STAR DYNAMOMETER — CFR ENGINE SETUP

Fig. 7



.018
.017
.016
.015
.014
.013
.012
.011
Mao, Lbs/sec
6
7
8
9
10
11
12
13
14
15
16
Inches of Water
.515" ASME ORIFICE
Flange Taps; 2" pipe
$M_a = M_{ao} \times \sqrt{\frac{P_1}{29.92} \times \frac{518.4}{T_1}}$

FIG. 8

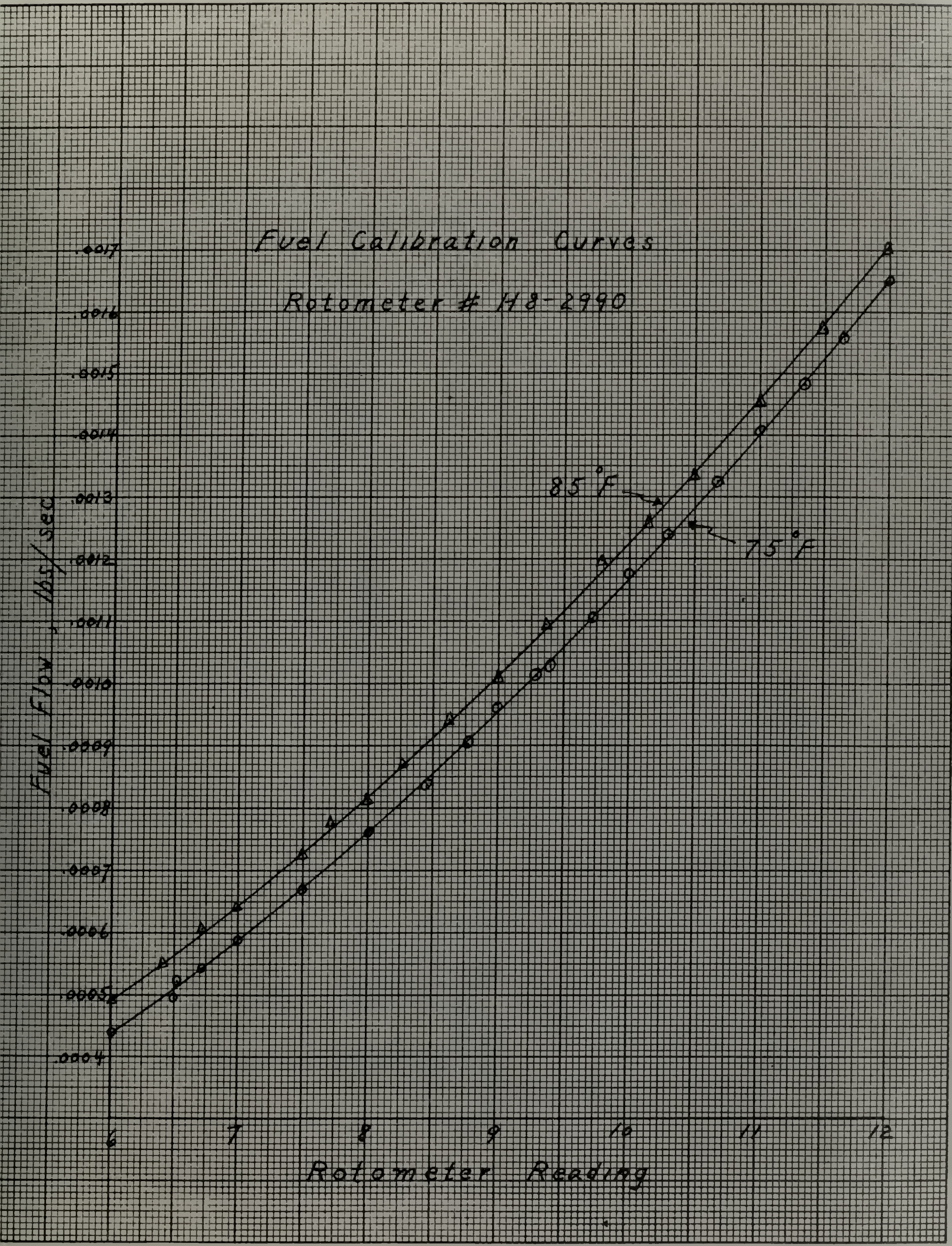
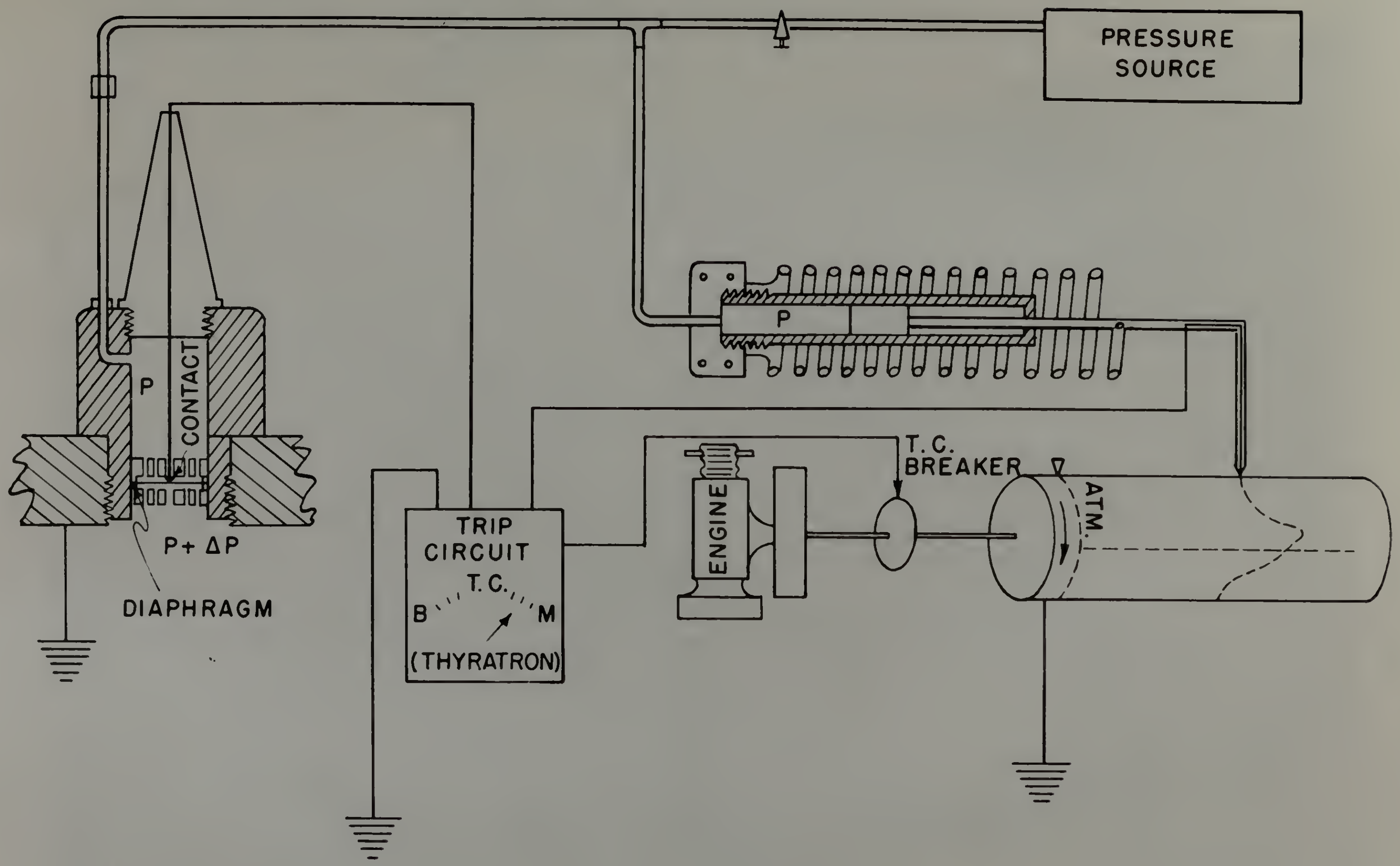

Fuel Calibration Curves
Rotometer # H8-2990
.0017
.0016
.0015
.0014
.0013
.0012
.0011
.0010
.0009
.0008
.0007
.0006
.0005
.0004
Fuel Flow, lbs/sec
85°F
75°F
6
7
8
9
10
11
12
Rotometer Reading

FIG. 9



HIGH SPEED INDICATOR

M. I. T.

Fig. 10

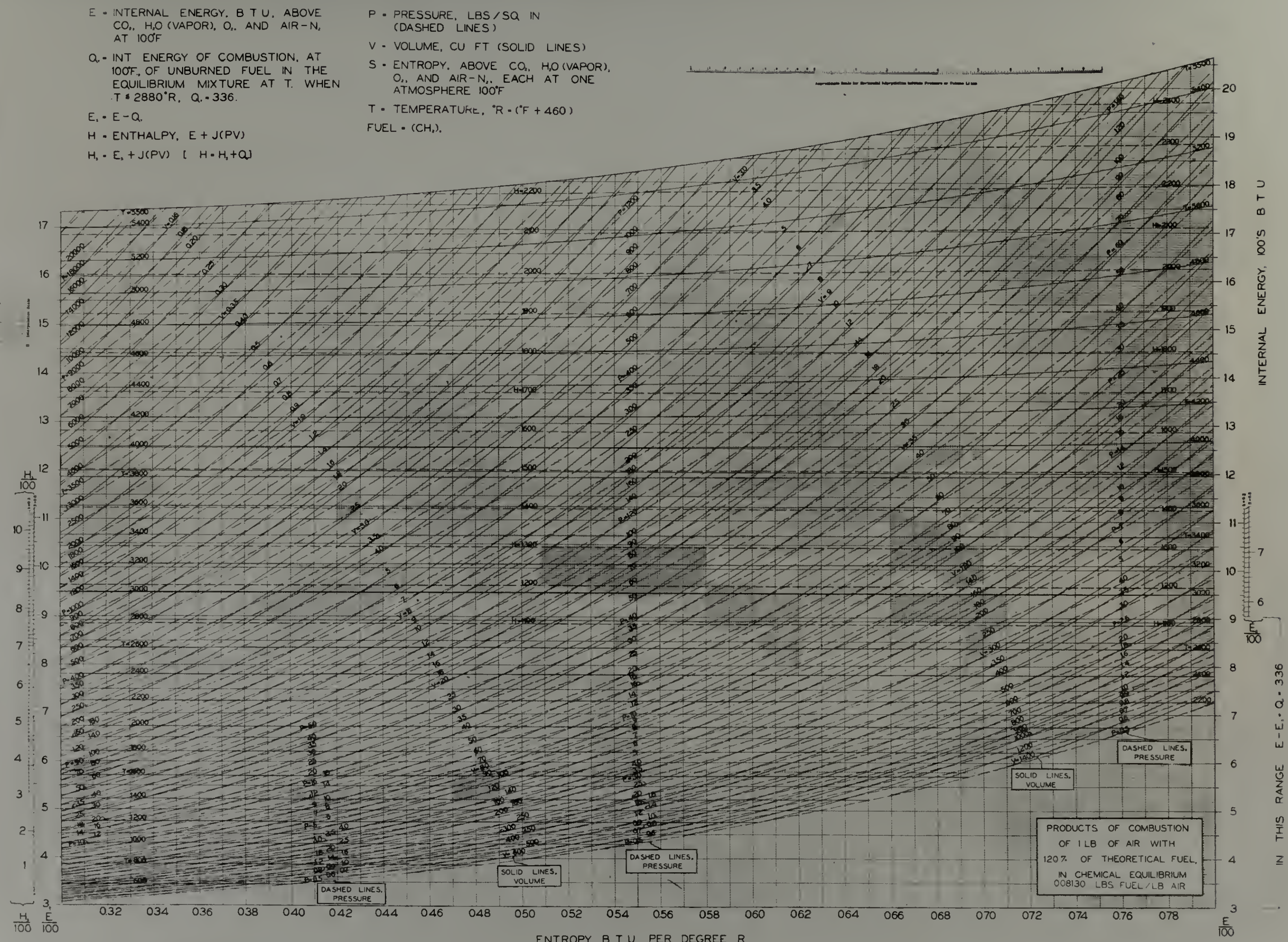
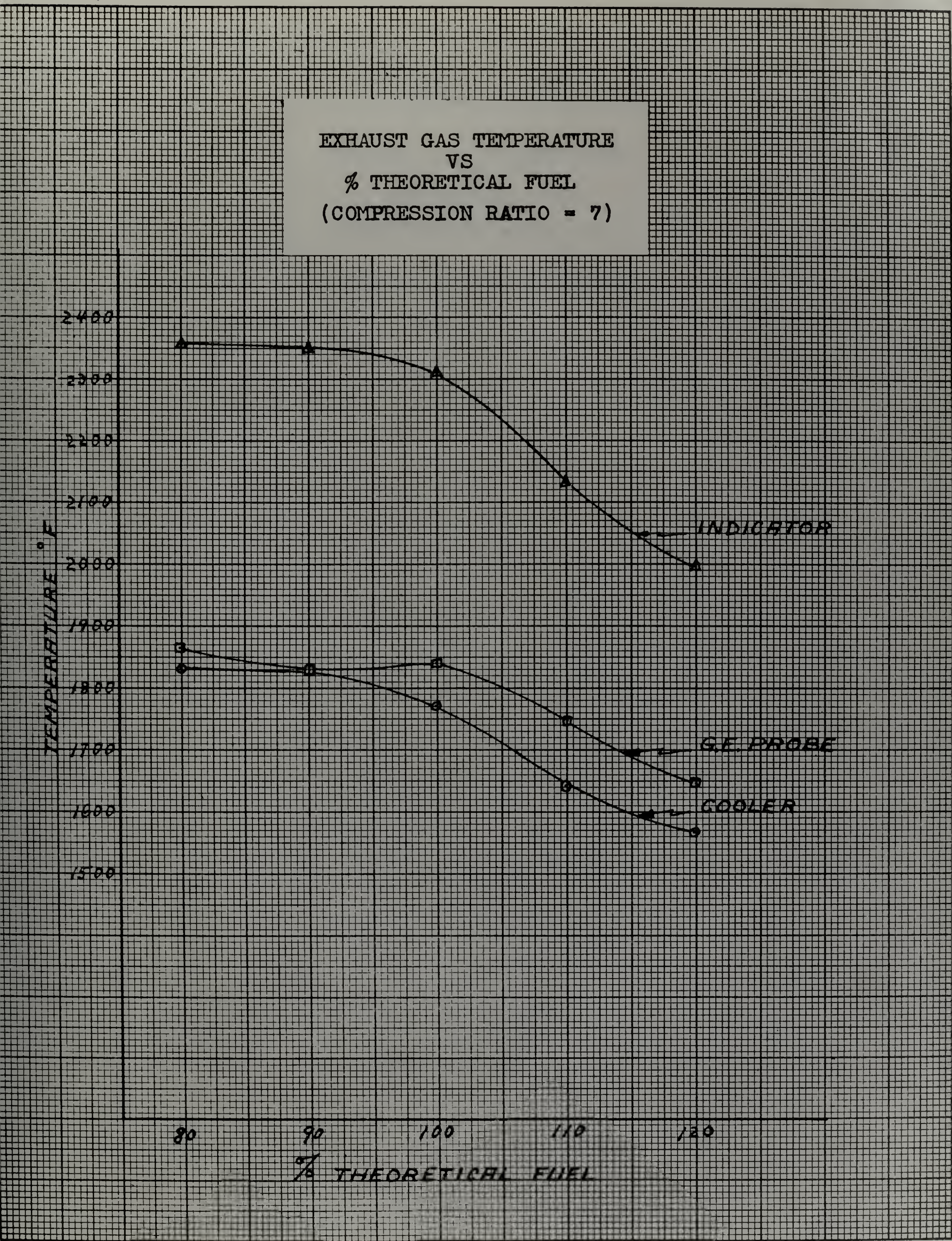
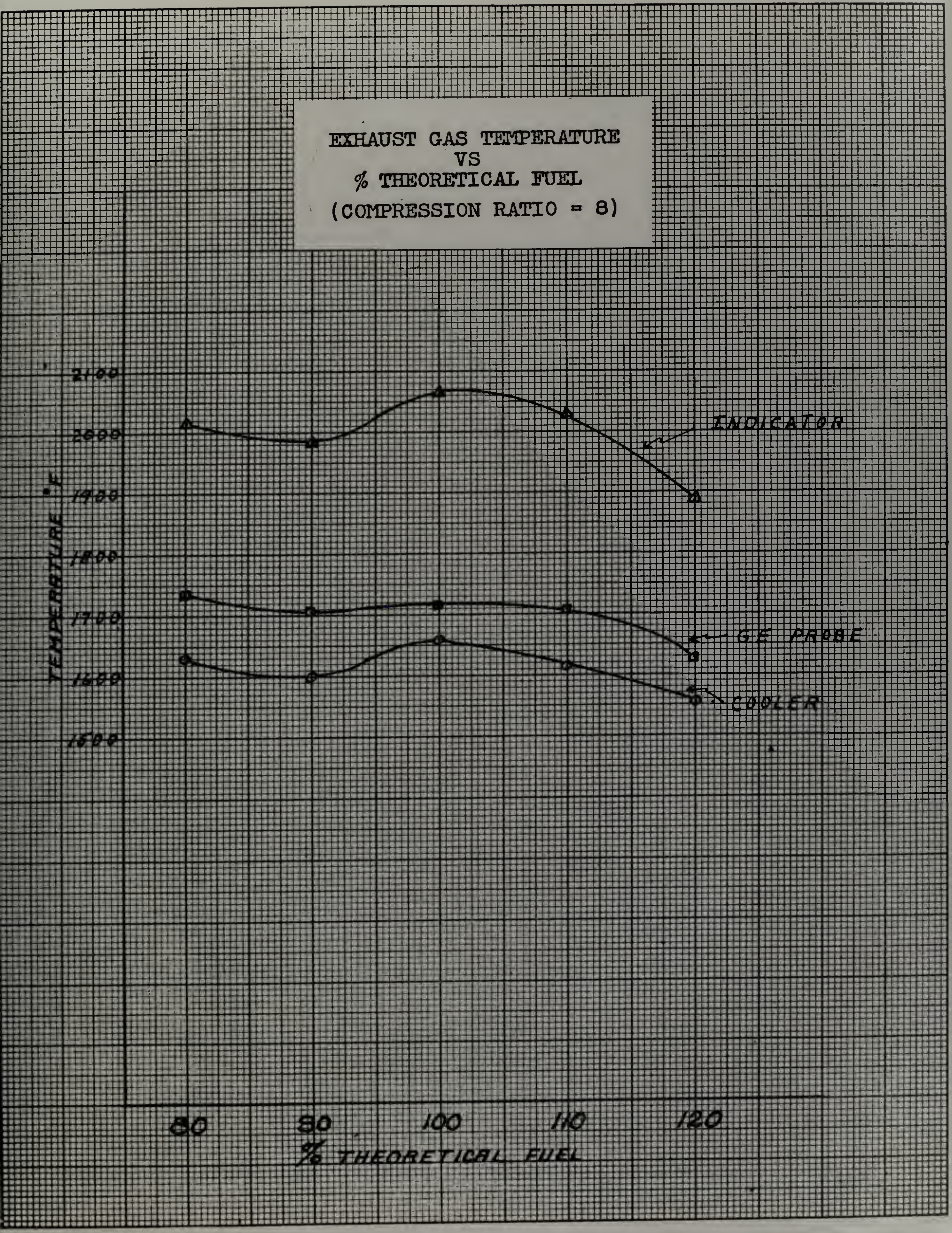

E = INTERNAL ENERGY, B T U, ABOVE CO₂, H₂O (VAPOR), O₂, AND AIR-N₂ AT 100°F
Q₀ = INT ENERGY OF COMBUSTION, AT 100°F, OF UNBURNED FUEL IN THE EQUILIBRIUM MIXTURE AT T. WHEN T ≤ 2880°R, Q₀ = 336.
E₁ = E − Q₀
H = ENTHALPY, E + J(PV)
H₁ = E₁ + J(PV) [H = H₁ + Q₀]
P = PRESSURE, LBS/SQ IN (DASHED LINES)
V = VOLUME, CU FT (SOLID LINES)
S = ENTROPY, ABOVE CO₂, H₂O (VAPOR), O₂, AND AIR-N₂, EACH AT ONE ATMOSPHERE 100°F
T = TEMPERATURE, °R = (°F + 460)
FUEL = (CH₂)ₙ
PRODUCTS OF COMBUSTION OF 1 LB OF AIR WITH 120% OF THEORETICAL FUEL, IN CHEMICAL EQUILIBRIUM 008130 LBS FUEL/LB AIR
DASHED LINES, PRESSURE
SOLID LINES, VOLUME
ENTROPY, B T U PER DEGREE R
INTERNAL ENERGY, 100'S B T U
IN THIS RANGE E − E₁ = Q₀ = 336
H₁/100
E/100
H/100
0.32
0.34
0.36
0.38
0.40
0.42
0.44
0.46
0.48
0.50
0.52
0.54
0.56
0.58
0.60
0.62
0.64
0.66
0.68
0.70
0.72
0.74
0.76
0.78
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
T=3500
H=2200
P=1200
V=30
M I T CAMBRIDGE, MASS MAY 1943

FIG. 11



EXHAUST GAS TEMPERATURE
VS
% THEORETICAL FUEL
(COMPRESSION RATIO = 7)

FIG. 12



EXHAUST GAS TEMPERATURE
VS
% THEORETICAL FUEL
(COMPRESSION RATIO = 8)

FIG. 13

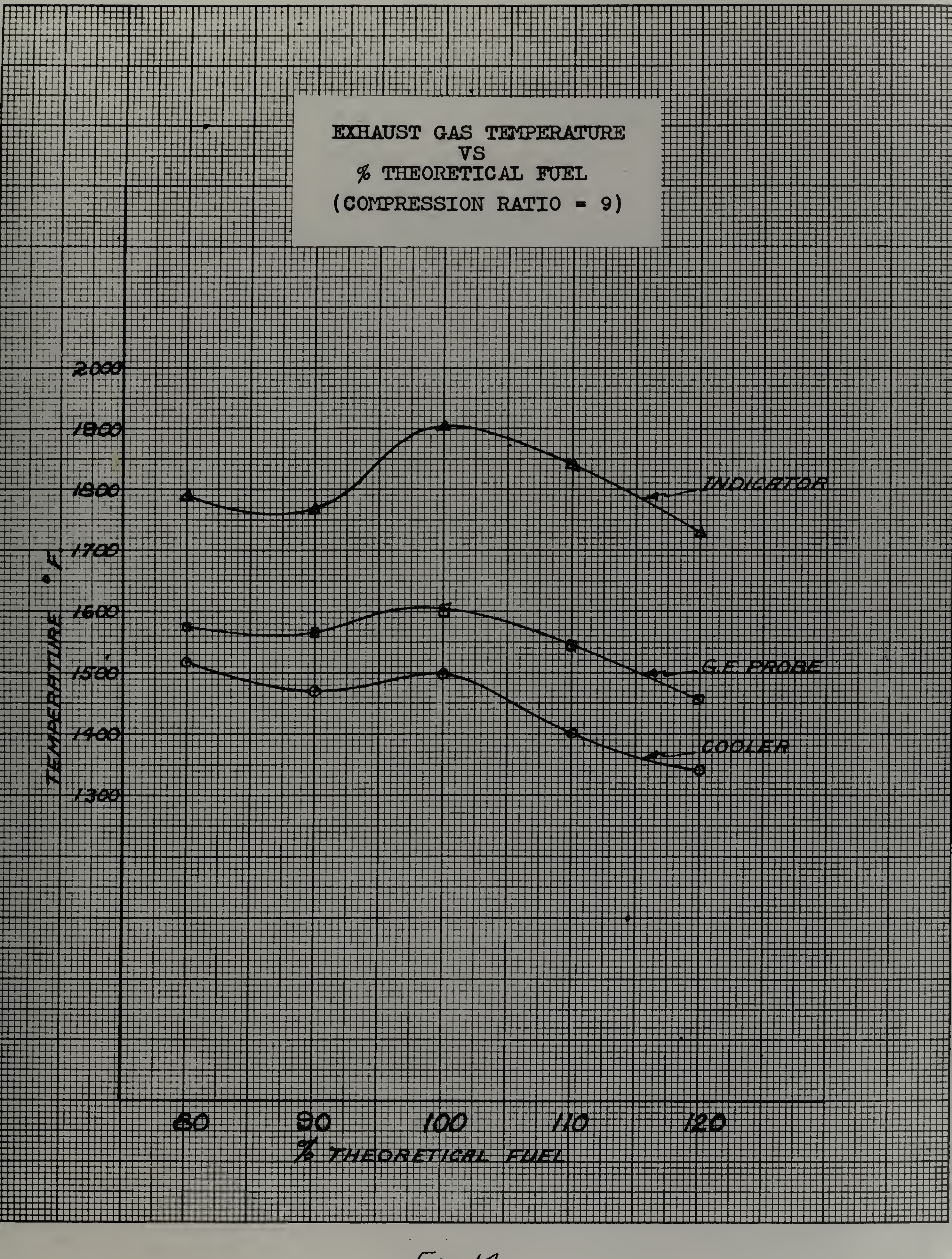

EXHAUST GAS TEMPERATURE
VS
% THEORETICAL FUEL
(COMPRESSION RATIO = 9)
TEMPERATURE °F
2000
1900
1800
1700
1600
1500
1400
1300
INDICATOR
G.E. PROBE
COOLER
80
90
100
110
120
% THEORETICAL FUEL

FIG. 14

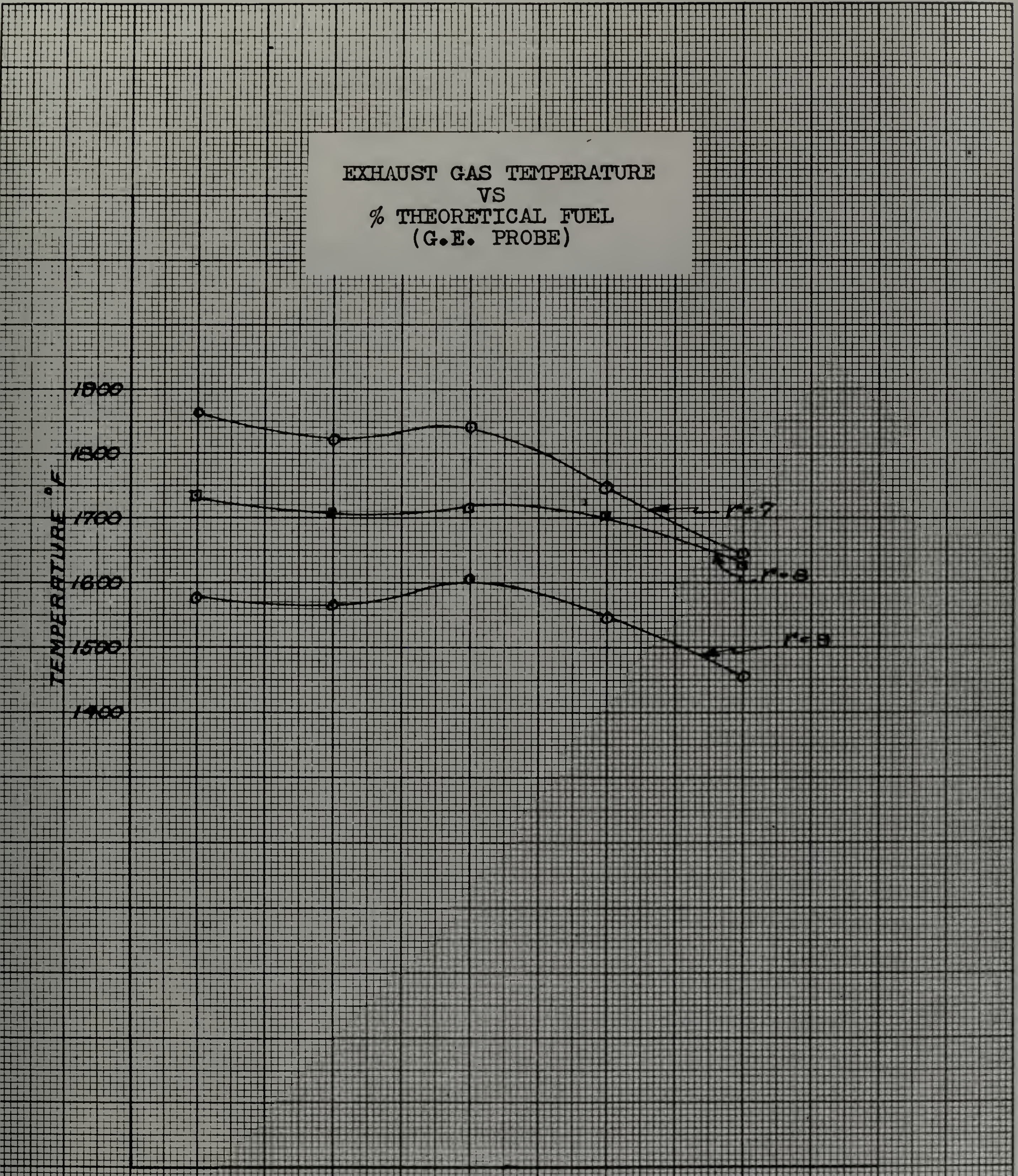

EXHAUST GAS TEMPERATURE
VS
% THEORETICAL FUEL
(G.E. PROBE)
1900
1800
1700
1600
1500
1400
TEMPERATURE °F
r=7
r=8
r=9
80
90
100
110
120
% THEORETICAL FUEL

FIG. 15

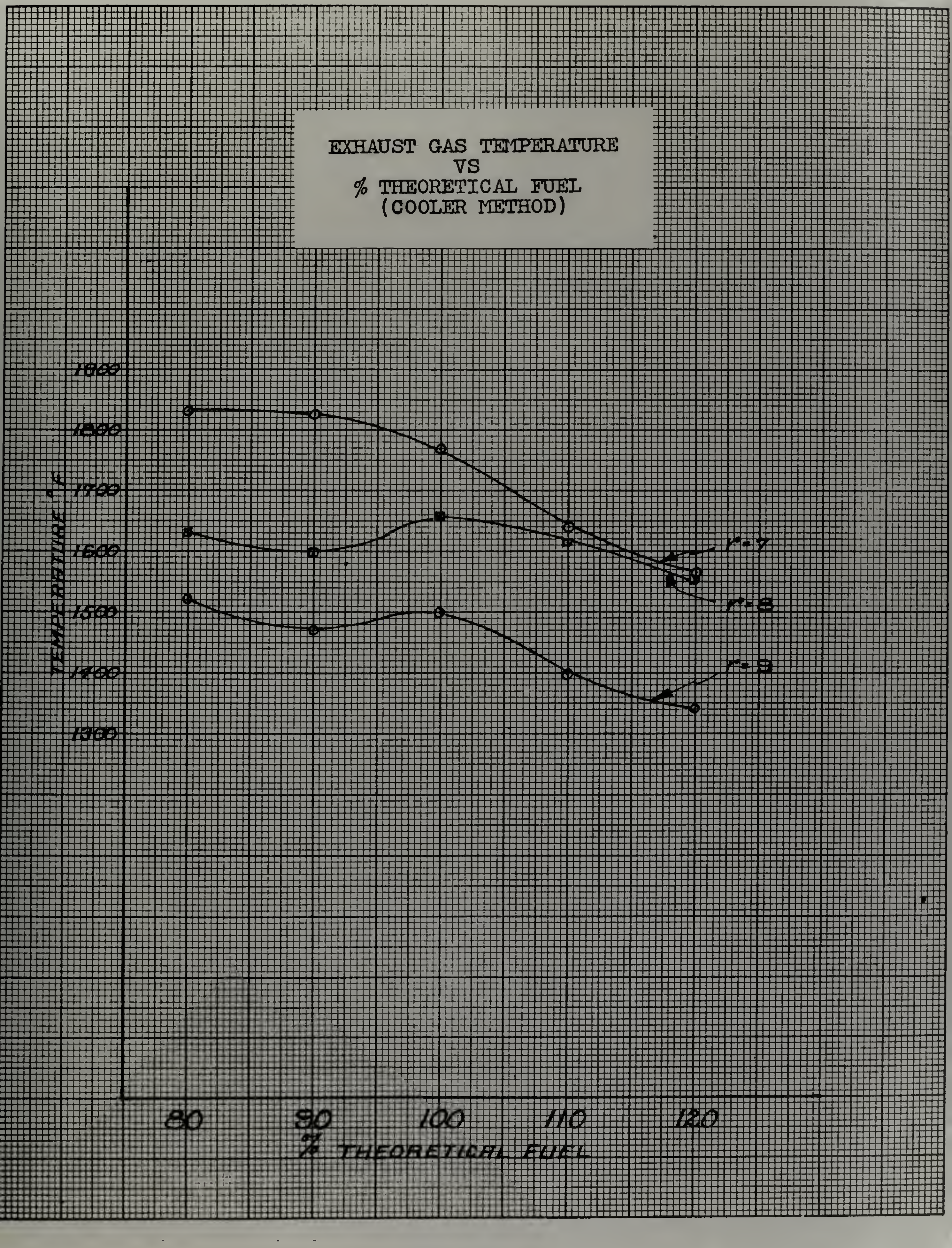

EXHAUST GAS TEMPERATURE
VS
% THEORETICAL FUEL
(COOLER METHOD)
1900
1800
1700
1600
1500
1400
1300
TEMPERATURE °F
80
90
100
110
120
% THEORETICAL FUEL
r=7
r=8
r=9

FIG. 16

EXHAUST GAS TEMPERATURE
VS
% THEORETICAL FUEL
(INDICATOR METHOD)

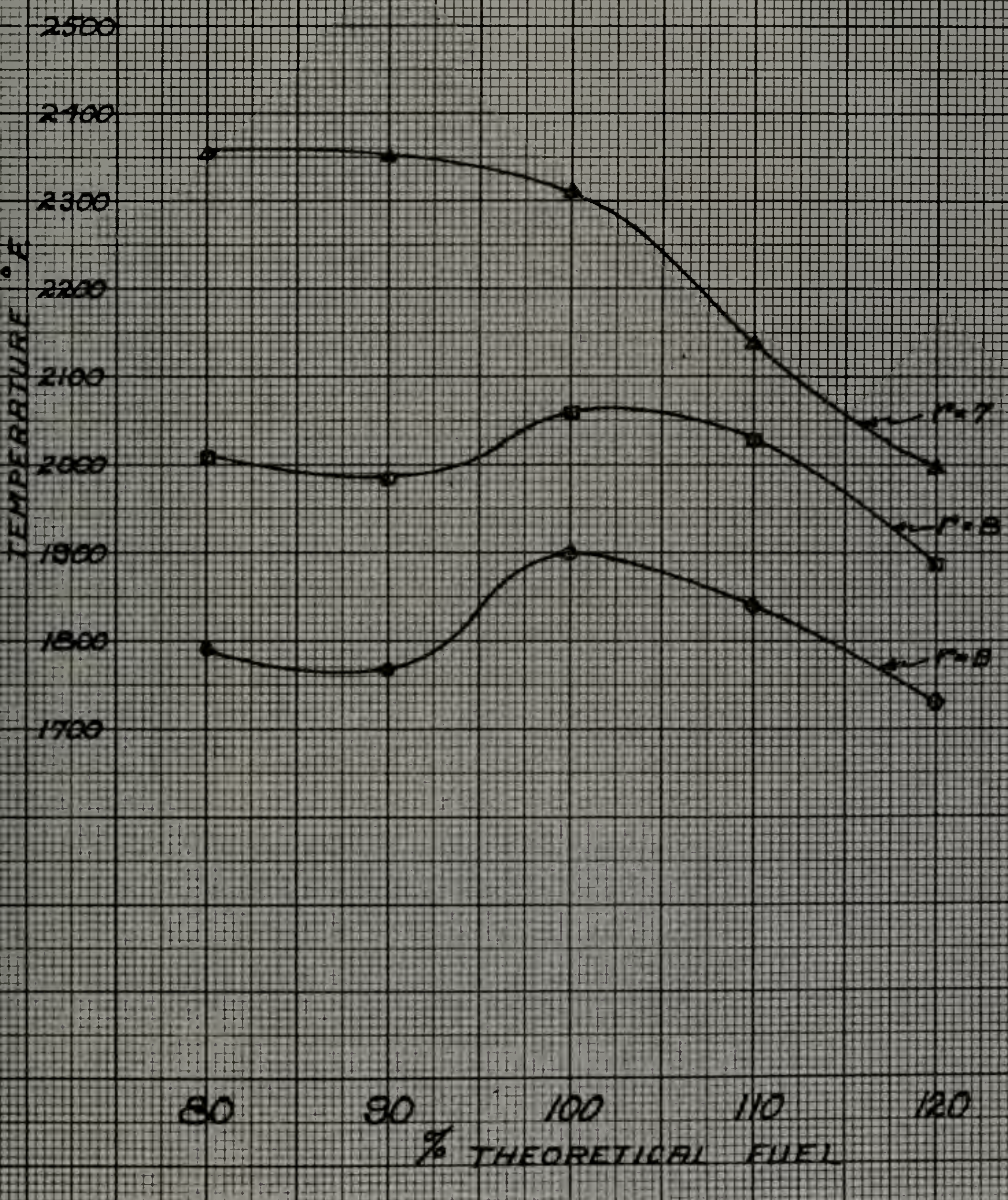


Fig. 17

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methods.

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